



Short communication

The effect of piston friction on the torsional natural frequency of a reciprocating engine

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Abstract

The torsional vibration of systems involving engines exhibits some phenomena not typical of vibration in general. These effects result from the geometry of the reciprocating mechanism and are revealed by detailed kinematic and dynamic analyses. Apart from the system's inertia varying with crank rotation, recent work by the authors has shown that it is also affected by piston-to-cylinder friction. In previous published work, that omitted friction, differences were observed between the measured and the predicted natural frequency function. This short paper revisits those differences and includes friction effects.

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1. Introduction

Due to the piston and conrod centre-of-mass positions changing their location with respect to the crankshaft axis, a single reciprocating mechanism has an effective rotary inertia which changes with crank angle. The inertia is a smoothly varying periodic function close to an average inertia with a superimposed second-order cosine. This twice-per revolution variation gives rise to sidebands on the main torsional natural frequency determined by the average inertia. Excitation at these side peaks can feed energy into the main natural frequency and cause, what in the industry is known as, 'secondary resonance'. Drew et al. [1] investigated the torsional excitation of variable inertia effects in a single cylinder reciprocating engine. They used both frequency and time domain models and presented what appeared to be the first experimental verification of the theory. A brief explanation of the rig used by Drew, shown in Fig. 1, and the test procedure is given here for completeness. The rig used a 120 cm³ Villiers engine with the flywheel, cylinder head, camshaft and timing gear removed. The simplified engine mechanism was motored via shafting supported in bearings. Using a servo drive it was possible to set the engine at a discrete crank angle and then torsionally excite the system about this position. A TAPTM accelerometer was located at the free end of the rig to measure the

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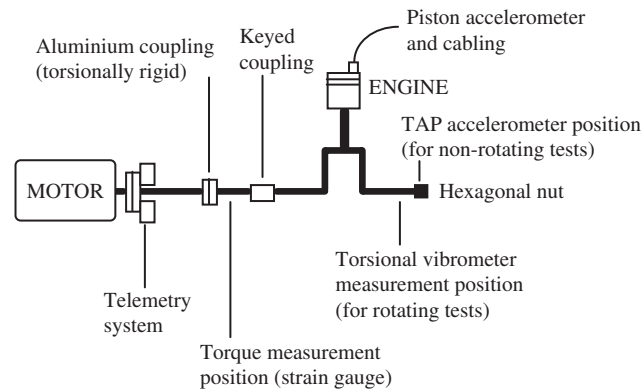


Fig. 1. Schematic diagram of the experimental rig used by Drew et al. [1].

angular response. At each crank angle sinusoidal excitation at different frequencies was applied and the natural frequency was determined from the largest response measured with the TAPTM. A receptance model [3] was used to predict the natural frequency function. Further details regarding set-up and physical data pertaining to the rig can be found in [1].

The measured and predicted results from this earlier work are shown in Fig. 3. Drew [1] observed a much greater frequency variation (15%) in the measured values than those predicted, with the greatest discrepancies occurring when the piston was located near mid-stroke. According to Drew the discrepancies between predicted and measured results could have been due to a number of reasons. These are paraphrased here:

1. The variation in natural frequency is very sensitive to the reciprocating mass value.
2. Slight errors in material properties may shift the node position resulting in a smaller contribution from the engine's inertia term in the predicted response.
3. Assumptions regarding the behaviour of the reciprocating mechanism are incorrect (backlash, bearing flexibility, etc.)

It was noted that a future investigation into these differences was necessary. The remainder of this paper revisits Drew's work and conducts a more detailed investigation.

2. Modelling

Guzzomi et al. [2] found that piston-to-cylinder friction influences the moment experienced by the engine block. Experiments and predictions showed that large discrepancies occur when friction is ignored; especially at low crank rotational speed. The study used the same engine as Drew [1]. Referring to Fig. 2, Guzzomi's analysis assumed that piston friction is dominated by two interactions: ring friction (F_r) produced by the static ring tension applying a distributed force against the cylinder wall, and dynamic friction (μS) produced by the piston side force (S) when the piston is translating up/down the cylinder. Ignoring the gap in the piston rings it is possible to group the distributed ring pack friction force, F_r , as shown in Fig. 2. If the piston is also assumed to remain collinear to the cylinder bore then both F_r and μS can be shown to effectively act on the piston crown. No combustion forces are included in Fig. 3.

The geometry of a reciprocating mechanism causes piston-to-cylinder friction and the piston side force to be interdependent. Recent work by the current authors has shown that piston friction may also affect an engine's effective inertia function. The inertia function with friction included is given by Eq. (1). It is not the purpose of this short paper to present the derivation of Eq. (1), it is instead to show the results obtained by including friction. A detailed mathematical analysis deriving Eq. (1) will be published shortly. The well-known equation for the inertia function without friction [1] is given by Eq. (2). Eq. (2) can also be

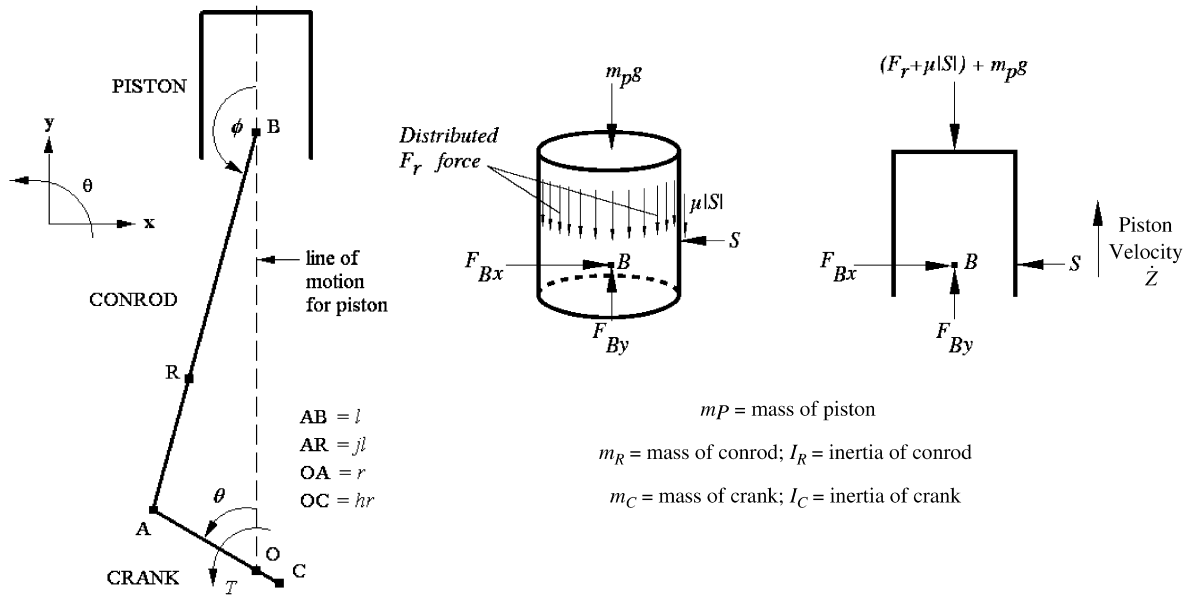
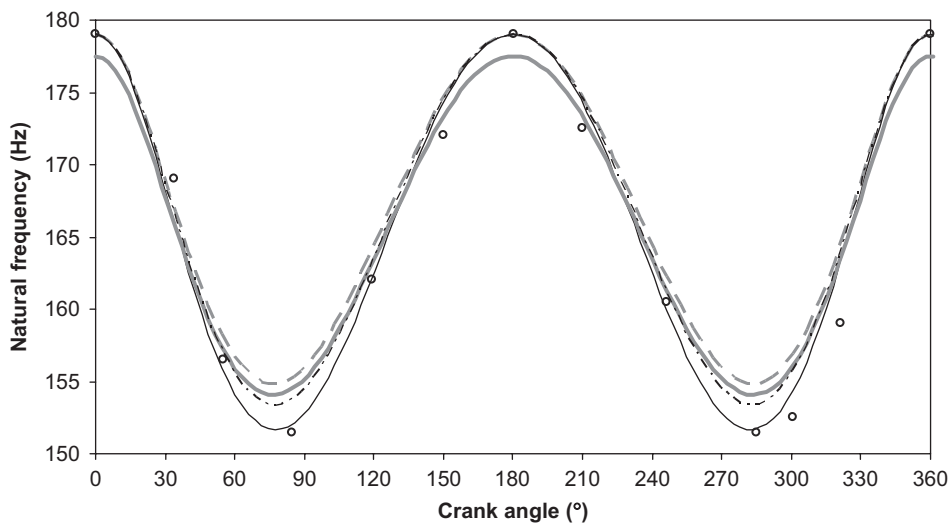


Fig. 2. Schematics of (a) engine mechanism [1] and (b) piston FBD [2].

Fig. 3. Torsional natural frequency as a function of crank angle: $\circ$, measured from [1]; —, predicted from [1]; — —, corrected system parameters, $\mu = 0$; - · - ·, corrected system parameters, $\mu = 0.3$; — — —, corrected system parameters, $\mu = 0.6$.

obtained by setting $\mu = 0$ in Eq. (1)

$$\begin{aligned}
 I_f(\theta) = & I_C + m_C h^2 r^2 + I_R H(\mu) \frac{r^2 \cos \theta}{(l \cos \phi)^2} \\
 & + m_P r^2 [H(\mu) \tan \phi (\cos \theta \tan \phi - \sin \theta) - \sin \theta (\cos \theta \tan \phi - \sin \theta)] \\
 & + m_R r^2 [(1-j) \cos^2 \theta - j \sin \theta \cos \theta \tan \phi + \sin^2 \theta] \\
 & + m_R r^2 j H(\mu) \left(\frac{j \cos \theta}{(\cos \phi)^2} - \sin \theta \tan \phi - \cos \theta \right),
 \end{aligned}$$

where

$$H(\mu) = \frac{\cos \theta + \mu \sin \theta}{1 + \mu \tan \phi} \begin{cases} \dot{z} \geq 0, S \geq 0, \\ \dot{z} < 0, S < 0, \end{cases}$$

$$H(\mu) = \frac{\cos \theta - \mu \sin \theta}{1 - \mu \tan \phi} \begin{cases} \dot{z} \geq 0, S < 0, \\ \dot{z} < 0, S \geq 0, \end{cases} \quad (1)$$

$$I(\theta) = I_C + m_C h^2 r^2 + I_R \left(\frac{r \cos \theta}{l \cos \phi} \right)^2 + m_P r^2 (\cos \theta \tan \phi - \sin \theta)^2$$

$$+ m_R r^2 [(1 - j)^2 \cos^2 \theta + (j \cos \theta \tan \phi - \sin \theta)^2]. \quad (2)$$

Friction must always oppose motion and since S may change sides during piston motion, the correct sign to use with the μ term is determined from both the sign of piston velocity and the sign of S [2]. S is a function of the crank rotational speed and any external piston loading term. In the absence of combustion, the external loading is due to the ring pack's friction force (F_r), which is dominated by the oil scraper ring [2], and dynamic piston friction.

3. Natural frequency function

The receptance program and the experimental rig used by Drew [1] were also available for this study. The system parameters used in Drew's model were checked and some small errors were discovered and corrected. The model was also modified to include friction. Drew's original results and the new predictions, with and without friction, are shown in Fig. 3. The experimental measurements are from a non-rotating engine and top dead centre (TDC) corresponds to zero degrees on the plots. The corrected parameters improve predictions at the DC positions, but discrepancies still remain when the piston is located near mid-stroke. Introducing friction into the model does not affect the predicted natural frequencies at the DC positions, because with no gudgeon pin offset or combustion S reduces to zero at these locations and hence dynamic friction, μS , is also zero. However, piston friction does have a significant effect when the piston is away from DC and S is non-zero. Results for $\mu = 0.3$ and 0.6 are shown in Fig. 3 and it is clearly evident that friction improves the predictions of the natural frequency function.

High values of μ may be expected for the experimental tests investigated. For oscillating tests with angular acceleration magnitudes of the magnitude noted in [1], minimal piston motion occurs and thus there is minimal opportunity to entrain lubricant. The Villiers engine also uses a splash lubrication system. Thus it is reasonable to assume that the lubrication was further compromised during the tests because the engine was not rotating. Furthermore, crank oscillation with zero mean speed causes similar piston-to-cylinder conditions as those experienced at DC positions under normal operating conditions. That is, the piston velocity changes sign, with the piston becoming momentarily stationary. It is well established [4,5] that this condition gives rise to high μ -values, since the lubrication regime approaches or becomes that of boundary lubrication.

4. Summary

Drew [1] presented experimental and predicted results for the varying natural frequency function of a single cylinder reciprocating engine, but noted differences between the predicted and experimental results. The largest discrepancies occurred near mid-stroke. In this paper those differences were investigated. New predictions drawing on recent work by the authors has resulted in a very good tie-up. Previously the error in the natural frequency function variation was 15%. Corrections to some parameter values resulted in the error decreasing to 12.5%. However mid-stroke discrepancies were still observed. A model including piston friction was then investigated. By including friction the error between measured and predicted natural frequencies was reduced further to 7% for $\mu = 0.3$ and 1% for $\mu = 0.6$. Reasons have been given as to why high μ -values may be expected for the test set-up used by Drew [1]. In an operating engine such high values of friction would not be expected because oscillations are superimposed on an average rotational speed. Thus, apart from the DC

locations, it is very unlikely for the direction of piston travel to reverse during operation. This means high sliding velocity and lubricant entrainment can be assumed. Also, the value of μ is highly dependent on variations in the lubrication regime and will not be constant for all crank positions [4]. In conclusion, the interdependence of the piston side force and piston-to-cylinder friction further contributes to the non-linear behaviour of engines. While this interdependence has been shown in the context of engines, the findings may extend to other mechanisms.

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